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Numerical Investigation of Thermal-Hydraulic Performance of a Flat Tube in Cross-Flow of Air

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***Abstract*—**Heat transfer from flat tube is studied numerically. Reynolds number is defined base on equivalent circular tube which is varied in range of 100 to 300. In these range of Reynolds number flow is considered to be laminar, unsteady, and incompressible. Equations are solved by using finite volume method. Results show that increasing *l*/D from 1 to 2 has insignificant effect on heat transfer and Nusselt number of flat tube is slightly lower than circular tube. However, thermal-hydraulic performance of flat tube is up to 2.7 times greater than circular tube.

***Keywords*—**Laminar flow, flat tube, convective heat transfer, heat exchanger.

I. Introduction

HEAT transfer from bluff bodies has many applications such as heat exchangers, air conditioning, refrigeration and so on. Zukauskas and Ziugzda [1], Zukauskas and Ulinskas [2], Zdravkovich [3], [4] published books about flow and heat transfer from circular tubes.

Lam et al [5] studied effects of wavy cylindrical tubes in a staggered tube bundle. They used experimental measurement and large eddy simulation technique. Their Reynolds number varied in range of 6800<Re<13400. Their results showed that by using wavy tube drag coefficient reduced and the fluctuating lift suppressed. Moawed [6] studied experimentally forced convention from outside surfaces of helical coiled tube. He studied ten helical coiled-tubes with various design parameter and Reynolds number in range of 6.6×10^2 to 2.3×10^3. He found that with small value of pitch ratio, higher average Nusselt number can be achieved. Simo Tala et al. [7] performed unsteady-RANS simulation to investigated effects of tube pattern on thermal-hydraulic characteristics in a two-rows finned-tube heat exchanger. They used three indicates that iso-sectional tube increases thermal-hydraulic performance of heat exchanger compared to classical finned-tube heat exchangers. Tan et al. [8] studied experimentally heat transfer and pressure drop performance of twisted oval tube heat exchanger. Their results show that heat transfer coefficient of the twisted tube is higher than smooth round tube with cost of some incensement of pressure drop.

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Nouri-Borujerdi and Lavasani [9]-[13] experimentally studied flow and heat transfer from a cam-shaped tube in cross-flow of air. Their results indicate that single cam-shaped tube performed better than circular tube. Lavasani and Bayat [14]-[16] numerically studied flow and heat transfer characteristics from two cam-shaped tubes in side-by-side and tandem arrangement. Mirabdolah Lavasani et al. [17] experimentally studied convective heat transfer from cam-shaped tube bank with in-line arrangement and Bayat et al. [18] experimentally studied thermal-hydraulic performance of cam-shaped tube bank in staggered arrangement. Their results show that thermal-hydraulic performance of cam-shaped tube bank in both in-line and staggered arrangement is about 5-6 times greater than circular tube bank.

Flat tubes due to lower air-side pressure drop and higher air side heat transfer compared to circular tube performed better in heat exchangers [19], [20]. In this study heat transfer from single flat tube with different aspect ratio in cross-flow of air is studied numerically.

II. Numerical Method

A. Problem Description and Governing Equation

The cross section profile of the flat tube is represented in Fig. 1. These tubes are comprised of two circles with two line segments tangent to them. Characteristic length for these tubes is the diameter of an equivalent circular cylinder, $D_{eq}=P/\pi$, whose circumferential length is equal to that of the flat tube. In this study flow characteristics of three different tubes is investigated which the dimensions of each of these tubes is presented in Table I.

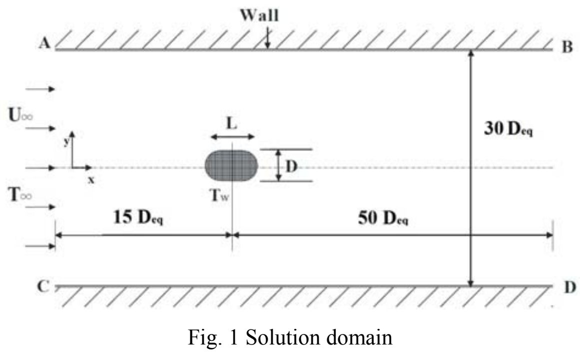

Fig. 1 Solution domain

TABLE I
DIMENSIONS OF FLAT TUBES

Tube No.	D *(cm)*	*l (cm)*	$D_{eq}=P/\pi$ *(cm)*
1	1	1	$1+2/\pi$
2	1	2	$1+4/\pi$

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The typical solution domain and the cylinder boundary definition and nomenclature used in this work are shown in Fig. 1. The inlet flow has a uniform velocity. The velocity range considered only covers laminar flow conditions. The solution domain is bounded by the inlet, the outlet, and by the plane confining walls, AB and CD. These are treated as solid walls, while AC and BD are the flow inlet and outlet planes.

In order to decrease the effect of entrance and outlet regions, the upstream and downstream lengths are 15D and 50D, respectively.

Equations are written for conservation of mass, momentum and energy in two dimensions. Cartesian velocity components U and V are used, and it has been assumed that the flow is steady and laminar, while the fluid is incompressible and Newtonian with constant thermal and transport properties. Furthermore, the effects of buoyancy and viscous dissipation are neglected. The governing equations consist of the following four equations for the dependent variables U, V, P and T:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

Equations (1) to (4) are the conservation of mass, x and y direction momentum equations and energy, respectively. The boundary conditions used for the solution domain shown in Fig. 2 are uniform inlet velocity, fully developed outflow and no-slip cylinder surface boundary. The total drag coefficient and Nusselt number of a flat tube are respectively as follows:

$$C_D=\frac{F_D}{0.5\rho U_\infty^2 D_{eq}} \tag{5}$$

$$Nu=\frac{qD}{Ak\,\Delta T} \tag{6}$$

where q is the total rate of heat transfer to the fluid and A is the total surface area of tubes. The temperature of the cylinders wall is 400 K and the bulk temperature of the cross-flow air is 300 K and ΔT is the difference between these temperatures.

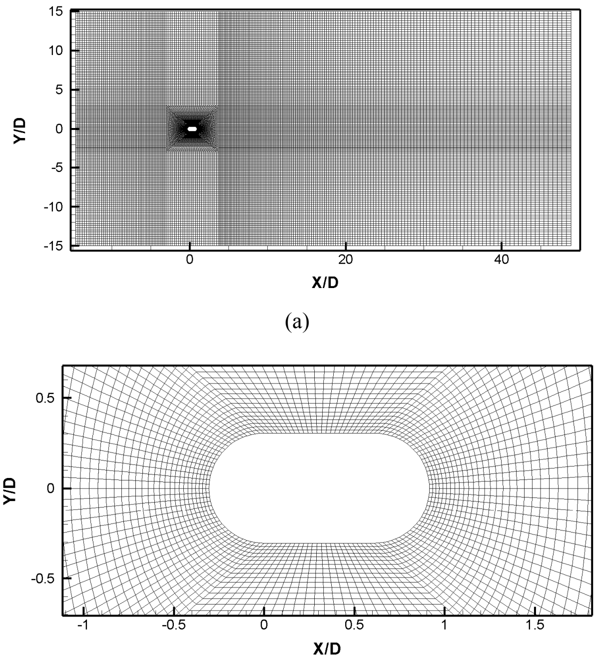
Thermal hydraulic performance of flat tube base on circular tube is defined by efficiency index η which has been proposed by Webb [21].

$$\eta=\frac{\bar{N}u_{ave.flat.}/\bar{N}u_{ave.cir.}}{C_{D,\,flat.}/C_{D,\,cir.}} \tag{7}$$

B. Numerical Method

This problem considers a 2D section of flat tube. For the simulations presented here, depending on the geometry used fine meshes of 35370 to 48234 elements were used. A sample of the mesh for the flat tube is shown in Fig. 2. In this domain quadrilateral cells are used in the regions surrounding the tube wall and the rest of the domain. In all simulation, a convergence criterion of 1×10^{-6} was used for all variables.

The second order upwind scheme was chosen for interpolation of the flow variables. The SIMPLE algorithm [22] has been adapted for the pressure velocity coupling. In all simulation, a convergence criterion of 1×10^{-6} was used for all variables.



(a)

(b)

Fig. 2 Computational grid: (a) entire computational domain, (b) closer view around tube

III. Result and Discussion

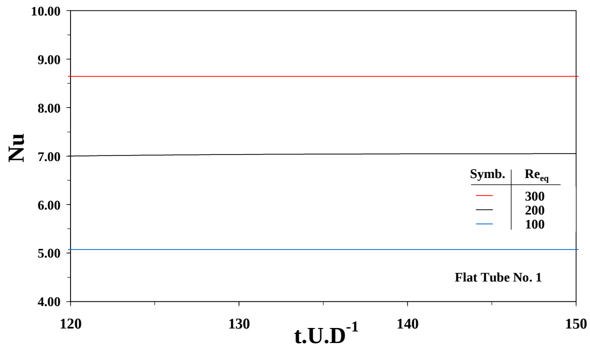
For the purpose of the validation of the solution procedure the present numerical simulations are first carried out for circular tube with constant temperature and its results is compared with other works in literature. Table II compares the Nusselt number of circular cylinder with the results of Zukauskas [1] and Churchill-Bernstein correlation [23].

TABLE II
COMPARISON OF NUSSELT NUMBER OF A CIRCULAR TUBE WITH ZUKAUSKAS & CHURCHIL-BERNSTEIN

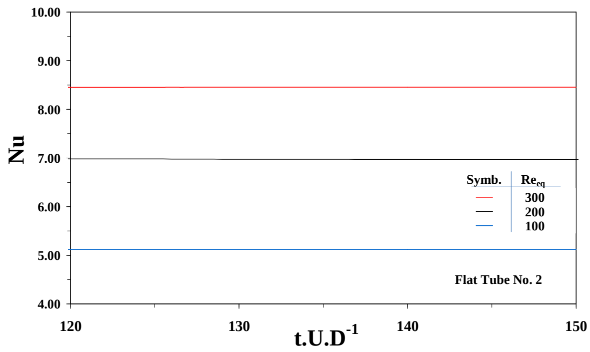
Author	Type of Data	***Reynolds Number***		
		100	***200***	***300***
Present work	Numercial	5.18	7.47	9.22
Zukauskas [1]	Experimental	5.18	7.15	8.64
Churchill and Bernstein [23]	------------	5.15	7.18	8.74

There is a difference of about 2-7% between the present results and the results of Zhukauskas and 1-5% with Churchill-Bernstein equation. It can therefore be concluded that the CFD code can be used to solve the flow field for similar geometries and conditions.

Fig. 4 shows variation of Nusselt number of both flat tubes with time and Fig. 5 represent variation of average Nusselt number with Reynolds number. Results show that by increasing Reynolds number from 100 to 300, Nusselt number increases about 70% and 65% for tube No.1 and 2, respectively. Moreover, for a fixed value of Reynolds number increasing *l*/D from 1 to 2 leads to insignificant changes in heat transfer and compare to circular tube, Nusselt number of flat tube with *l*/D=1 and 2 is about 1-8% lower.



(a)



(b)

Fig. 3 Times histories of Nusselt number: (a) $l/D=1$ (b) $l/D=2$

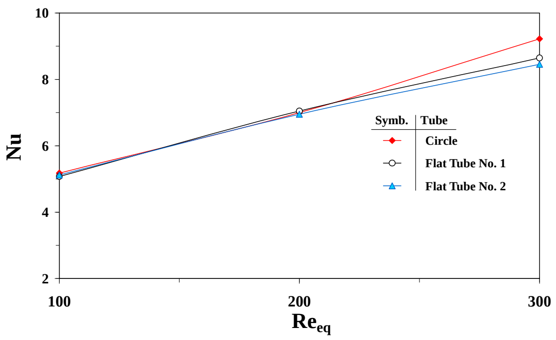


Fig. 4 Heat transfer from cam-shaped tube

Comparison of Nu/C_D of flat tube with circular tube is presented in Fig. 5. Drag coefficient is much lower than circular tube, therefore, for all rang of Reynolds number Nu/C_D of flat tube is about 67 to 169 percent greater than circular tube with equivalent diameter.

Fig. 6 represents thermal-hydraulic performance of both flat tubes with Reynolds number. Results show that thermal-hydraulic performance of both flat tubes is higher than circular tube due to lower drag coefficient of these tubes compare to circular one. Thermal-hydraulic performance of flat tube No. 1 and tube No .2 is about 1.7-2 times and 2-2.7 times greater than circular tube, respectively. As a result, using these tubes in heat exchanger will lead to increasing thermal performance and decreasing pressure drop.

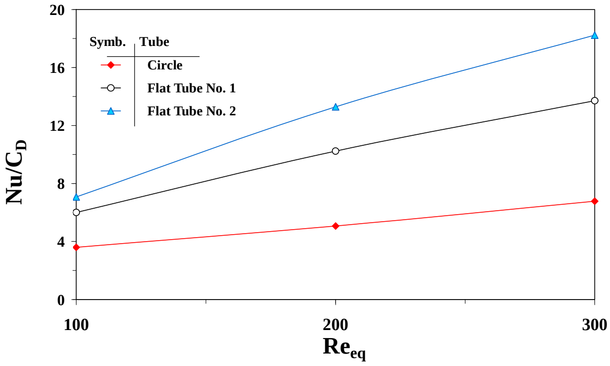


Fig. 5 Comparison of Nu/C_D of flat tube with circular tube

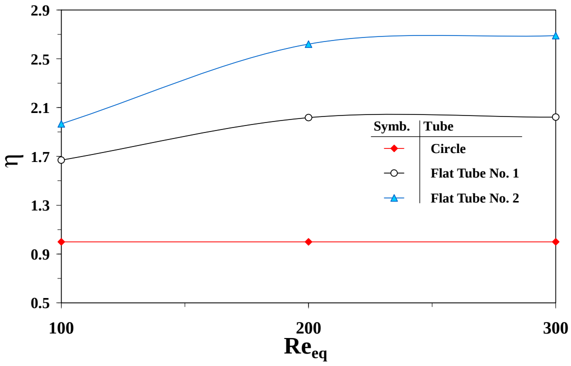


Fig. 6 Thermal-hydraulic performance of flat tube

IV. Conclusion

Unsteady heat transfer from flat tube with constant wall temperature is investigated numerically by using finite volume method. Reynolds number varied in range of 100 to 300. Results show that by increasing Reynolds number from 100 to 300 heat transfer from flat tube increases up to 70%. Heat transfer of flat tube is about 1-8% lower than circular tube with equivalent diameter. However, thermal-hydraulic performance of this tube is up to 2.7 times greater than circular tube.

NOMENCLATURE

D	Diameter
k	thermal conductivity, $(W.m^{-1}.K^{-1})$
ℓ	Distance between centers, (m)
P	Pressure, circumferential length
Re	Reynolds number, $U_\infty D_{eq}/\upsilon$
Nu	Nusselt number
T	Temperature, (K)
t	time (s)
u	x-direction velocity, (m/s)
v	y-direction velocity, (m/s)

Greek

ρ	Density, $(kg.m^{-3})$
υ	fluid kinematic viscosity, $(m^2.s^{-1})$
η	Thermal- Hydraulic performance

Subscripts

Flat	flat tube
Cir	Circular tube
eq	Equivalent
∞	Free stream

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